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I, ANNA MAIJA MADL, ACTING TEAM LEADER EXAMINATION SUPPORT & SALES hereby certify that annexed is a true copy of the Provisional specification in connection with Application No. PP 9959 for a patent by VENTRASSIST PTY LTD AND UNIVERSITY OF TECHNOLOGY, SYDNEY filed on 23 April 1999.

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Fifth day of May 2000

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# AUSTRALIA

## Patents Act 1990

## PROVISIONAL SPECIFICATION

Invention Title: ***Non-Contact Estimation and Control System***

The invention is described in the following statement:

Our Ref: 991008

# NON-CONTACT ESTIMATION & CONTROL SYSTEM

The present invention relates to a non-contact estimation and control system and, more particularly but not exclusively, to such a system suitable for use with a blood pump and, even more particularly, to such a pump having a hydrodynamically suspended impeller.

## BACKGROUND

It is desirable in some environments in which pumps are utilised that the control systems used to control the pumps rely on sensors which do not protrude into or otherwise invade or make direct mechanical contact with the fluid being pumped. The term "non-contact" is used to describe the nature of such sensing systems.

With particular reference to pumps for the pumping of blood to aid the heart it is desirable that any sensing systems be "non-contact" because any invasion of the blood being pumped can give rise to unwanted forces, stresses or other impediments to the flow of the blood leading to damage to the blood.

There are some non-contact sensors such as laser or ultra-sonic based devices. However, these devices add complexity, cost, size, bulk and tend to increase the probability of failure of the overall system of which they form a part.

It is therefore desirable if required control input variables could be derived from devices forming part of the pump or pump drive system so that additional, single purpose sensors are not required.

Also, whilst it is considered desirable to have non-contact sensing systems for control of the pumping of such fluids it has, in practice, been difficult to achieve this

outcome because of a high degree of unpredictability of the control variables, for example as between flow rate predicted and actual flow rate.

It is an object of at least some embodiments of the present invention to overcome or ameliorate the abovementioned problems.

**BRIEF DESCRIPTION OF INVENTION**

Accordingly, in one broad form of the invention there is provided an estimation and control system for a pump; said pump of the type having an impeller located within a pump cavity in a pump housing; said housing having a fluid inlet in fluid communication with said cavity; said housing having a fluid outlet in fluid communication with said pump cavity; said impeller urged to rotate about an impeller axis so as to cause fluid to be urged from said inlet through said pump cavity to said pump outlet; said impeller urged to rotate by impeller urging means; said impeller supported for rotational movement by impeller support means; said impeller maintained at or near a predetermined speed of rotation by control means acting on said impeller urging means; said control means receiving as input variables a first input variable comprising power consumed by said urging means; said control means receiving a second input variable comprising actual speed of rotation of said impeller; said control means thereby estimating head or rate of flow of said fluid to an approximation of predetermined accuracy relying on signals available from said urging means.

Preferably said pump has a substantially constant head versus flow rate characteristic over a predetermined flow rate range.

Preferably the blades of said impeller are such that a midline chord angle of said blades is inclined substantially radially to internal walls of said pump cavity.

Preferably said system relies on sensing of EMF induced in one or more coils forming part of said urging means.

Preferably said impeller include blades inclined such that relative velocity of fluid off-flow from said blades is substantially radial with respect to said impeller axis.

Preferably said pump includes impeller support means which is sufficiently adaptive to allow repositioning of said impeller in use to conserve energy as a function of fluid viscosity.

Preferably said pump is a low specific speed pump.

Preferably said pump has a specific speed in the range 100-2000 rev/min $(gal/min)^{1/2}ft^{-3/4}$.

More preferably said pump has a specific speed of approximately 900-1000 rev/min $(gal/min)^{1/2}ft^{-3/4}$.

In a further broad form of the invention there is provided in combination a rotary blood pump and an estimation and control system therefor, said pump having an impeller suspended hydrodynamically within a pump housing by thrust forces generated by the impeller during movement in use of the impeller as it rotates about an impeller axis; said estimation and control system of the type described above.

In yet a further broad form of the invention there is provided an estimation and control system for a pump; said pump of the type having an impeller located within a pump cavity in a pump housing; said housing having a fluid inlet in fluid communication with said cavity; said housing having a fluid outlet in fluid communication with said pump cavity; said impeller urged to rotate about an impeller axis so as to cause fluid to be urged from said inlet through said pump

cavity to said pump outlet; said impeller urged to rotate by impeller urging means; said impeller supported for rotational movement by impeller support means; said pump maintained at or near a predetermined operating point by control means acting on said impeller urging means; said control means receiving as input variables at least a first input variable derived from said urging means; said control means receiving at least a second input variable also derived from said urging means; said control means thereby calculating an estimate of said operating point to an approximation of predetermined accuracy relying on signals available from said urging means; said control means controlling said pump by comparing said predetermined operating point with said estimate of said operating point.

**BRIEF DESCRIPTION OF DRAWINGS**

Embodiments of the invention will now be described with reference to the accompanying drawings wherein:

Fig. 1 is a block diagram of a non-contact estimation and control system in accordance with a first embodiment of the invention applied to a blood pump;

Fig. 2 illustrates an impeller of the blood pump of Fig. 1;

Fig. 3 is a head versus flow characteristic for the pump of Fig. 1;

Fig. 4 is a characteristic estimation curve utilised by the non-contact estimation and control system of Fig. 1;

Fig. 5 is a side, cut-away view of the pump of Fig. 1;

Fig. 6 is a plan, cut-away view of the coil and magnet system of the pump of Fig. 1;

Fig. 7 is an exploded, perspective view of a centrifugal pump assembly according to a further embodiment of the invention;

Fig. 8 is a perspective view of the impeller of the assembly of Fig. 7;

Fig. 9 is a perspective, cut away view of the impeller of Fig. 2 within the pump assembly of Fig. 7;

Fig. 10 is a side section indicative view of the impeller of Fig. 8;

Fig. 11 is a detailed view in side section of edge portions of the impeller of Fig. 10;

Fig. 12 is a block diagram of an electronic driver circuit for the pump assembly of Fig. 7;

Fig. 13 is a graph of head versus flow for the pump assembly of Fig. 7;

Fig. 14 is a graph of pump efficiency versus flow for the pump assembly of Fig. 7;

Fig. 15 is a graph of electrical power consumption versus flow for the pump assembly of Fig. 7;

Fig. 16 is a plan, section view of a variation of the pump assembly of Fig. 7 showing a volute arrangement according to a further preferred embodiment;

Fig. 17 is a plan, section view of an alternative pump assembly showing an alternative volute arrangement;

Fig. 18 is a plan view of an impeller according to a further embodiment of the invention;

Fig. 19 is a plan view of an impeller according to a further embodiment of the invention;

Fig. 20 illustrates efficiency versus specific speed for a range of pump types, various ones of which are suitable to be controlled by the non-contact estimation and control system of the present invention.

## DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

Embodiments of the present invention relate to a non-contact estimation and control system usable, although not exclusively, with blood pump systems of the type illustrated in Fig. 1.

In this instance the estimation and control system 10 operates on and receives sensor feedback from pump assembly 11 adapted for implantation in human body 12 and arranged to operate in parallel across at least a part of heart 13 so as to at least assist if not fully take over the pumping function of heart 13.

The pump assembly 11 includes an impeller 14 having vanes 15 which, when urged to rotate by a magnetic field generated in one or more of coils 16, 17 generates a pressure head H across the pump assembly 11 and causes a flow of blood Q therethrough. In this instance the impeller 14 is both a radial pump impeller and a rotor of motor 18 by virtue of the inclusion of magnets (not shown) within at least part of the impeller 14.

Monitoring means 19 is adapted to sense electric current appearing in one or more of coils 16, 17 via sensing line 35 which, in conjunction with monitoring of voltage derived from commutation controller 32 (which injects current into one or more of the same coils 16, 17) permits the monitoring means 19 to derive power input ($P_{in}$) consumed by motor 18 and actual rate of rotation of the motor/impeller 14 ($n_a$).

By means of equation 1.1 (in Fig. 1) it is thereby possible for monitoring means 19 to calculate an estimation of flow Q (and/or head H) for input into microprocessor 20. Microprocessor 20 accepts these estimates and, together with other desired set points and predetermined values calculates

a desired set motor speed $n_{set}$ which commutation controller 32 accepts via line 33. The commutation controller 32 then injects current into one or more of coils 16, 17 in order to cause impeller 14 to rotate at that set (desired) speed.

Fig. 2 illustrates the impeller 14 in greater detail.

Fig. 3 illustrates the head versus flow characteristic achievable with the impeller of Fig. 2 for a number of different motor powers (Pin). Fig. 4 illustrates the characteristic curve used by the monitoring means 19 for example 1 (to follow) in accordance with the equation 1.1.

Example 1

Flow rate and pressure difference (or head) are key variables needed in the control of implantable rotary blood pumps. However, use of flow and/or pressure probes can decrease reliability and increase system power consumption and expense. For given fluid viscosity, the flow state is determined by any two of the four pump variables: flow, pressure difference, speed and electromagnetic torque (apart from the possibility of non uniqueness of solutions). Instead of torque, motor current or input power can be used. Thus if viscosity is known, or if its influence is sufficiently small, flow rate and pressure difference can be estimated from the motor speed and input power, which can be determined from current and voltage measurements on the motor input leads.

The centrifugal blood pumps of Figs. 1-6, which uses a hydrodynamic bearing and constructed so that, the variation with viscosity is sufficiently small to enable flow and pressure difference estimation using signals derived from the coils 16, 17.

A mock loop was setup (see fig 1 ) consisting of the

pump and 2.4m of 3/8″ tubing giving a net fluid volume of 177ml.

The fluid filled tubing was sunk into a water bath with controlled heater. Temperature sensors were attached to the tubing to provide visual feedback on fluid temperature. Pressure taps were made on the inlet and outlet nozzles of the pump which interfaced to a differential pressure transducer with digital display to measure pressure across the pump. A Clamp on Transonics flow probe and meter were used to measure flow rate and input power (motor supply voltage and current) was monitored via digital panel meters on the power supply. Pressure was varied by adjustment of a tubing clamp and motor speed by varying applied motor voltage.

Two tests were conducted. The first with 5% saline, the second with red blood cell suspensions, haematocrit being 32%. In both cases the circulating fluids were heated to 37°C. 5% saline was chosen since its viscosity is about that of water at 23deg C. Also Transonics Systems recommend this as a reference fluid, giving a flow reading 3% higher than expected.

25 points flow rate, pressure head, pump speed and electrical input power were measured for both fluids.

Data for saline and blood was combined and correlated on a surface plot describing both flow rate as a function of motor speed and input power as illustrated in Fig. 4.

Curve fitting of this plot produced the equation $Q=20.29+4.73\ln(Pin)-0.55\sqrt{(n)}$ where Q is flow rate in L/min, Pin is electrical input power to the motor in watts and n is motor speed in rpm. The maximum error for this prediction was 4% for the combined data. Pressure head across the pump

was described by the relationship $\Delta P = -13.68 - 6.59\ln(Pin) + 2.18e\text{-}5\ (n)^2$ with equivalent accuracy. Two different rotor designs have been tested in this manner to date both yielding similar accuracy curve fits of the form $Q=a+b.\ln(Pin)+c.\sqrt{(n)}$ and of the form $\Delta P = a+b.\ln(Pin)+c.(n)^2$.
The viscosity of saline is approximately 1mPas. The Viscosity of blood (Hct = 32%) given pump shear rates of greater than $100s^{-1}$ is near 3 mPas. Blood viscosity varies from approximately 2.4 to 4.5 mPas over the physiological range in question for shear rates greater than $100s^{-1}$. The variation in viscosity from 1 to 3mPas produced a maximum error of 4% in the prediction of flow rate.

The pump of Figs. 1-6 has characteristics such that the model for flow rate prediction based on motor input power and speed is not greatly affected by variation in viscosity. This suggests for this pump it is possible to determine flow without using a separate flow sensor with acceptable accuracy.

The reason for low error in prediction given change in viscosity are postulated as follows: Firstly that the "flat" H-Q curves for this pump giving small variation in pressure head for given flow rates. Secondly the nature of the hydrodynamic bearing. Although the pump has relatively high disc friction forces, which tend to be most sensitive to viscosity changes, the rotor in this case conserves energy by repositioning in free space according to the fluid viscosity. Thirdly, the size, where surface roughness is relatively smaller than for smaller higher speed pumps.

Fig. 5 illustrates the pump assembly 11 in cross section as utilised with example 1.

Fig. 6 illustrates in cross section the coil and magnet arrangement used in conjunction with example 1.

With reference to Figs. 2, 5 and 6 iron yokes are placed outside the coils to increase the magnetic flux and hence increase motor efficiency, and also to reduce stray magnetic fields in the body. The yokes are positioned so that the axial magnetic force on the impeller is zero when it is central in the housing cavity. Furthermore, the yokes are placed at considerable distances from the impeller to keep the negative magnetic stiffness sufficiently low that is places only a small additional demand on the hydrodynamic suspension when the impeller shifts away from the cavity mid-position.

Given the large distance to the yokes, a slotless winding and axisymmetric yokes were chosen. The use of axisymmetric yokes implies zero "cogging" torque. The winding topology coil chosen is of "second harmonic" type with just three coils, one per phase, in each of the body and cover windings. Fig. 6 depicts the cover winding. The body windings align axially with the cover windings but must be bent in several directions to avoid the volute and inlet. This second harmonic topology avoids coil overlaps and is consequently neat and compact and give low copper mass. However, it is less efficient than other winding options with greater coil mass.

The efficiency is increased by tilting the magnet alignment to an angle of 22.5° from the pump axis (as indicated in Fig. 1 by the magnet hatching), intermediate between the 45° conical body and the flat cover. The cover coil and axial flux form an axial flux motor, and the body coil and flux are intermediate between an axial and radial flux motor.

The motor can be driven by a six-step, sensorless commutation inverter. Superimposed over the coils in Fig. 6 are magnets at an instant when the currents are switched from phases a and c conducting to phases b and c conducting (or v.v.). Parallel coil connection of the cover and body coils (each connected in star configuration) enables some redundancy, in that the motor still runs with the loss of a coil.

The materials used were Ti-6A1-4V for the housing and impeller shell, high remanence NdFeB magnets (VACODYM 510 HR) embedded in the impeller, iron for the yokes (mild steel in prototypes but to be laminated silicon steel) and varnished copper wire for the coils.

The measured negative magnetic stiffness of the teardrop impeller is -4000 N/m (± 10%). The axial clearance gaps are 0.1mm when the impeller is central (this is to match a 0.05mm taper on the blades for thrust generation so that after a shift of 0.05mm, the thrust forces are maximal from one impeller face and negligibly small from the other face). Thus if the impeller is shifted axially by the full amount possible (as at start-up if axis vertical), then the magnetic force on the impeller is 0.4 N force. This is less than the impeller weight of 46 gforce, and is considered acceptable. If the yokes are any closer, the force would be higher, increasing the risk of touchdown. Similarly, if the clearance gaps were increased to slacken manufacturing tolerances, then the maximal magnetic force would be increased.

The measured motor efficiency is between 45% and 48% curves, for speeds between 2000 rpm and 2500 rpm and motor output power between 3 and 7W. For example, at 2250 rpm and 3 W motor output (roughly rated conditions), the copper loss

was 1.7W, the eddy loss in the titanium was 1.0 W, and the iron loss in mild steel yokes was 0.7 W, giving a motor efficiency of 47%.

With reference to Figs. 7 to 15 inclusive there is shown a further preferred embodiment of a pump assembly 200 incorporating an estimating and control system of the type described with reference to Figs. 1-6.

With particular reference initially to Fig. 7 the pump assembly 200 comprises a housing body 201 adapted for bolted connection to a housing cover 202 and so as to define a centrifugal pump cavity 203 therewithin.

The cavity 203 houses an impeller 204 adapted to receive magnets 205 within cavities 206 defined within blades 207. As for the first embodiment the blades 207 are supported from a support cone 208.

Exterior to the cavity 203 but forming part of the pump assembly 200 there is located a body winding 209 symmetrically mounted around inlet 210 and housed between the housing body 201 and a body yoke 211.

Also forming part of the pump assembly 200 and also mounted external to pump cavity 203 is cover winding 212 located within winding cavity 213 which, in turn, is located within housing cover 202 and closed by cover yoke 214.

The windings 212 and 209 are supplied from the electronic controller of Fig. 12. As for the first embodiment the windings are arranged to receive a three phase electrical supply and so as to set up a rotating magnetic field within cavity 203 which exerts a torque on magnets 205 within the impeller 204 so as to urge the impeller 204 to rotate substantially about central axis TT of cavity 203 and in line with the longitudinal axis of inlet 210. The

impeller 204 is caused to rotate so as to urge fluid (in this case blood liquid) around volute 215 and through outlet 216.

The assembly is bolted together in the manner indicated by screws 217. The yokes 211, 214 are held in place by fasteners 218. Alternatively, press fitting is possible provided sufficient integrity of seal can be maintained.

Fig. 8 shows the impeller 204 of this embodiment and clearly shows the support cone 208 from which the blades 207 extend. The axial cavity 219 which is arranged, in use, to be aligned with the longitudinal axis of inlet 210 and through which blood is received for urging by blades 207 is clearly visible.

The cutaway view of Fig. 9 shows the axial cavity 219 and also the magnet cavities 206 located within each blade 207. The preferred cone structure 220 extending from housing cover 202 aligned with the axis of inlet 210 and axial cavity 219 of impeller 204 is also shown.

Fig. 10 is a side section, indicative view of the impeller 204 defining the orientations of central axis FF, top taper edge DD and bottom taper edge BB, which tapers are illustrated in Fig. 11 in side section view.

Fig. 11A is a section of a blade 207 of impeller 204 taken through plane DD as defined in Fig. 10 and shows the top edge 221 to be profiled from a leading edge 223 to a trailing edge 224 as follows: central portion 227 comprises an ellipse having a semi-major axis of radius 113mm and a semi-minor axis of radius 80mm subtended on either side by a region of no radius and then followed by leading conical surface 225 and trailing conical surface 226 on either side thereof as illustrated in Fig. 11A.

The leading edge 223 is radiused as illustrated.

Fig. 11B illustrates in cross-section the bottom edge 222 of blade 207 cut along plane BB of Fig. 10.

The bottom edge includes cap 228 utilised for sealing magnet 205 within cavity 206.

In this instance substantially the entire edge comprises a straight taper with a radius of 0.05mm at leading edge 229 and a radius of 0.25mm at trailing edge 230.

The blade 207 is 6mm in width excluding the radii at either end.

Fig. 12 comprises a block diagram of the electrical controller suitable for driving the pump assembly 200 and comprises a three phase commutation controller 232 adapted to drive the windings 209, 212 of the pump assembly. The commutation controller 232 determines relative phase and frequency values for driving the windings with reference to set point speed input 233 derived from physiological controller 234 which, in turn, receives control inputs 235 comprising motor current input and motor speed (derived from the commutation controller 232).

Fig. 13 is a graph of pressure against flow for the pump assembly 200 where the fluid pumped is 18% glycerol for impeller rotation velocity over the range 1500 RPM to 2500 RPM. The 18% glycerol liquid is believed to be a good analogue for blood under certain circumstances.

Fig. 14 graphs pump efficiency against flow for the same fluid over the same speed ranges as for Fig. 13.

Fig. 15 is a graph of electrical power consumption against flow for the same fluid over the same speed ranges as for Fig. 13.

FURTHER EMBODIMENTS

In the forms thus far described top surfaces of the blades 8, 207 are angled at approximately $45^0$ with respect to the longitudinal axis of the impeller 100, 204 and arranged for rotation with respect to the internal walls of a similarly angled conical pump housing. The top surfaces are deformed so as to create the-necessary restriction in the gap between the top surfaces of the blades and the internal walls of the conical pump housing thereby to generate a thrust which can be resolved to both radial and axial components.

In the examples thus far the bottom faces of the blades 207 comprise surfaces substantially lying in a plane at right angles to the axis of rotation of the impeller and, with their deformities define a gap with respect to a lower inside face of the pump housing against which a substantially only axial thrust is generated.

Other arrangements are possible which will also, relying on these principles, provide the necessary balanced radial and axial forces. Such arrangements can include a double cone arrangement where the conical top surface of the blades is mirrored in a corresponding bottom conical surface. The only concern with this arrangement is the increased depth of pump which can be a problem for in vivo applications where size minimisation is an important criteria.

With reference to Fig. 18 a further embodiment of the invention is illustrated comprising a plan view of the impeller 300 forming part of a "channel" pump. In this embodiment the blades 301 have been widened relative to the blades 207 of the third embodiment to the point where they are almost sector-shaped and the flow gaps between adjacent blades 301, as a result, take the form of a channel 302, all in communication with axial cavity 303.

A further modification of this arrangement is illustrated in Fig. 19 wherein impeller 304 includes sector-shaped blades 305 having curved leading and trailing portions 306, 307 respectively thereby defining channels 308 having fluted exit portions 309.

As with the earlier embodiments the radial and axial hydrodynamic forces are generated by appropriate profiling of the top and bottom faces of the blades 301, 305 (not shown in Figs. 18 and 19).

**SUMMARY OF OPERATION PRINCIPLES**

The estimation and control system described with reference to the previous embodiments is "sensorless" in that it derives an estimate of relevant pump parameters from signals available from one or more of the drive coils of the motor. Hence no separate sensor device is required to control the pump assembly in use.

It is hypothesized that the ability to control the pump assembly in this manner to a sufficiently good approximation derives from shaping the impeller of the pump so that a relatively flat head versus flow characteristic is obtained over the flow rate range expected and/or required of the pump, in use.

It is postulated that relative radial off-flow and lack of constraint of the fluid within the impeller derived from the relatively low number of impeller blades aids in achieving the relatively flat pump characteristic curves as shown for example in Figs. 3 and 13.

It is also postulated that, in the embodiments described in the specification, the impeller blades are arranged to guide fluid carefully through the rotor so as to reduce re-circulation. There are also relatively large gaps between

the blades so that the fluid is relatively poorly constrained leading to loosely constrained flow of fluid within the pump housing.

## CLAIMS

1. An estimation and control system for a pump; said pump of the type having an impeller located within a pump cavity in a pump housing; said housing having a fluid inlet in fluid communication with said cavity; said housing having a fluid outlet in fluid communication with said pump cavity; said impeller urged to rotate about an impeller axis so as to cause fluid to be urged from said inlet through said pump cavity to said pump outlet; said impeller urged to rotate by impeller urging means; said impeller supported for rotational movement by impeller support means; said impeller maintained at or near a predetermined speed of rotation by control means acting on said impeller urging means; said control means receiving as input variables a first input variable comprising power consumed by said urging means; said control means receiving a second input variable comprising actual speed of rotation of said impeller; said control means thereby estimating head or rate of flow of said fluid to an approximation of predetermined accuracy relying on signals available from said urging means.
2. The system of Claim 1 wherein said pump has a substantially constant head versus flow rate characteristic over a predetermined flow rate range.
3. The system of Claim 1 wherein blades of said impeller are such that a midline chord angle of said blades is inclined substantially radially to internal walls of said pump cavity.
4. The system of Claim 1 which relies on sensing of EMF induced in one or more coils forming part of said urging means.

5. The system of Claim 1 wherein said impeller includes blades inclined such that relative velocity of fluid off-flow from said blades is substantially radial with respect to said impeller axis.

6. The system of Claim 1 having impeller support means which is sufficiently adaptive to allow repositioning of said impeller in use to conserve energy as a function of fluid viscosity.

7. The system of Claim 1 wherein said pump is a low specific speed pump.

8. The system of Claim 7 wherein said pump has a specific speed in the range 100-2000 rev/min $(\mathrm{gal/min})^{1/2}\mathrm{ft}^{-3/4}$.

9. The system of Claim 7 wherein said pump has a specific speed of approximately 900-1000 rev/min $(\mathrm{gal/min})^{1/2}\mathrm{ft}^{-3/4}$.

10. In combination a rotary blood pump and an estimation and control system therefor, said pump having an impeller suspended hydrodynamically within a pump housing by thrust forces generated by the impeller during movement in use of the impeller as it rotates about an impeller axis; said estimation and control system of the type claimed in any one of claims 1 to 9.

11. The blood pump of claim 10 wherein said thrust forces are generated by blades of said impeller.

12. The blood pump of claim 11 wherein said thrust forces are generated by edges of said blades of said impeller.

13. The blood pump of claim 12 wherein said edges of said blades are tapered or non-planar, so that a thrust is created between the edges and the pump housing during relative movement therebetween.

14. The blood pump of claim 11 wherein said edges of said blades are shaped such that the gap at the leading edge

of the blade is greater than at the trailing edge and thus the fluid which is drawn through the gap experiences a wedge shaped restriction which generates a thrust.

15. The blood pump of claim 10 wherein the pump is of centrifugal type or mixed flow type with blades of said impeller open on both front and back faces of the pump housing.

16. The blood pump of claim 15 wherein the front face of the pump housing is made conical, in order that the thrust force perpendicular to the conical surface has a radial component, which provides a radial restoring force to a radial displacement of the impeller axis during use.

17. The blood pump of claim 11 wherein the driving torque of said impeller derives from the magnetic interaction between permanent magnets within the blades of the impeller and oscillating currents in windings encapsulated in the pump housing.

18. The rotary blood pump of claim 10 when dependent from any one of claims 1 to 6 wherein said pump is of axial type.

19. The rotary blood pump of claim 18 wherein within a uniform cylindrical section of the pump housing, said impeller includes tapered blade edges which form a radial hydrodynamic bearing.

20. The rotary blood pump of claim 19 wherein an interior of the pump housing is made with reducing radius at the two ends, and wherein the end hydrodynamic thrust forces have an axial component which can provide the axial bearing.

21. The rotary blood pump of claim 19 wherein magnetic forces provide the axial bearing.

22. A rotary blood pump having a housing within which an impeller acts by rotation about an impeller axis to cause a pressure differential between an inlet side of the pump housing of said pump and an outlet side of the pump housing of said pump; said impeller suspended hydrodynamically by thrust forces generated by the impeller during movement in use of the impeller; said pump controlled by the estimation and control system of any one of claims 1 to 9.

23. The pump of claim 22 wherein said impeller includes magnetic material therein, the magnetic material encapsulated within a biocompatible shell or coating.

24. The pump of claim 23 wherein said biocompatible shell or coating comprises of a material which can be applied at low temperature such as a diamond coating.

25. The pump of claim 22 wherein internal walls of said pump which can come into contact with said blades during use are coated with a hard material such as titanium nitride or diamond coating.

26. A seal-less, shaft-less pump comprising a housing defining a chamber therein and having a liquid inlet to said chamber and a liquid outlet from said chamber; said pump further including an impeller located within said chamber; the arrangement between said impeller, said inlet, said outlet and the internal walls of said chamber being such that upon rotation of said impeller about an impeller axis relative to said housing, liquid is urged from said inlet through said chamber to said outlet; and wherein thrust forces are generated by said

impeller; said pump controlled by the estimation and control system of any one of claims 1 to 9.

27. The pump of claim 26 wherein said thrust forces are generated by blades of said impeller.

28. The pump of claim 27 wherein said thrust forces are generated by edges of said blades of said impeller.

29. The pump of claim 28 wherein said edges of said blades are tapered or non-planar.

30. The pump of any one of claim 28 wherein said edges of said blades are shaped such that a gap at the leading edge of each of said blades is greater than at a trailing edge thereof whereby fluid which is drawn through the gap experiences a wedge shaped restriction which generates a thrust relative to said housing.

31. The pump of claim 27 wherein the pump is of centrifugal type or mixed flow type with said blades of said impeller open on both front and back faces of the pump housing.

32. The pump of claim 31 wherein the front face of the pump housing is made conical, in order that the thrust perpendicular to its conical surface at any point has a radial component, which provides a radial restoring force to a radial displacement of the impeller axis.

33. The pump of claim 31 wherein the driving torque of said impeller derives from the magnetic interaction between permanent magnets within the blades of the impeller and oscillating currents in windings encapsulated in the pump housing.

34. The pump of claim 26 wherein said pump is of axial type.

35. The pump of claim 34 wherein within a uniform cylindrical section of the pump housing, tapered blade edges form a radial hydrodynamic bearing.

36. The pump of claim 34 wherein the pump housing is made with reducing radius at opposed ends, and wherein the end hydrodynamic thrust forces have an axial component which can provide the axial bearing.

37. The pump of claim 34 wherein magnetic forces or other means can provide the axial bearing.

38. A pump having a housing within which an impeller acts by rotation about an axis to cause a pressure differential between an inlet side of a housing of said pump and an outlet side of the housing of said pump; said impeller suspended hydrodynamically in at least one of a radial or axial direction by thrust forces generated by the impeller during movement in use of the impeller; said pump controlled by the estimation and control system of any one of claims 1 to 9.

39. The pump of claim 38 wherein said impeller includes magnetic material therein, the magnetic material encapsulated within a biocompatible shell or coating.

40. The pump of claim 39 wherein said biocompatible shell or coating comprises a diamond coating.

41. The pump of claim 39 wherein internal walls of said pump which can come into contact with said impeller during use are coated with a hard material such as titanium nitride or diamond coating.

42. The pump of claim 38 wherein at least upper and lower surfaces of blades of said impeller are interconnected by a structure having deformities in the outer surfaces thereof so that a thrust is created between said

surfaces and the adjacent pump casing during relative movement therebetween.

43. A method of hydrodynamically suspending and controlling an impeller within a rotary pump for support in at least one of a radial or axial direction; said method comprising incorporating a deformed surface in at least part of said impeller so that, in use, a thrust is created between said deformed surface and the adjacent pump casing during relative movement therebetween.

44. The method of claim 43 wherein said deformed surface includes a taper.

45. The method of claim 44 wherein said taper is arranged so that there is a larger gap at a leading edge thereof between said impeller and said pump casing than at a trailing edge thereof.

46. An estimation and control system for a pump; said pump of the type having an impeller located within a pump cavity in a pump housing; said housing having a fluid inlet in fluid communication with said cavity; said housing having a fluid outlet in fluid communication with said pump cavity; said impeller urged to rotate about an impeller axis so as to cause fluid to be urged from said inlet through said pump cavity to said pump outlet; said impeller urged to rotate by impeller urging means; said impeller supported for rotational movement by impeller support means; said pump maintained at or near a predetermined operating point by control means acting on said impeller urging means; said control means receiving as input variables at least a first input variable derived from said urging means; said control means receiving at least a second input variable also

derived from said urging means; said control means thereby calculating an estimate of said operating point to an approximation of predetermined accuracy relying on signals available from said urging means; said control means controlling said pump by comparing said predetermined operating point with said estimate of said operating point.

$$Q \triangleq f\left[P_{in} \text{ and } \sqrt{n_a}\right] \quad \text{--- } 1.1$$

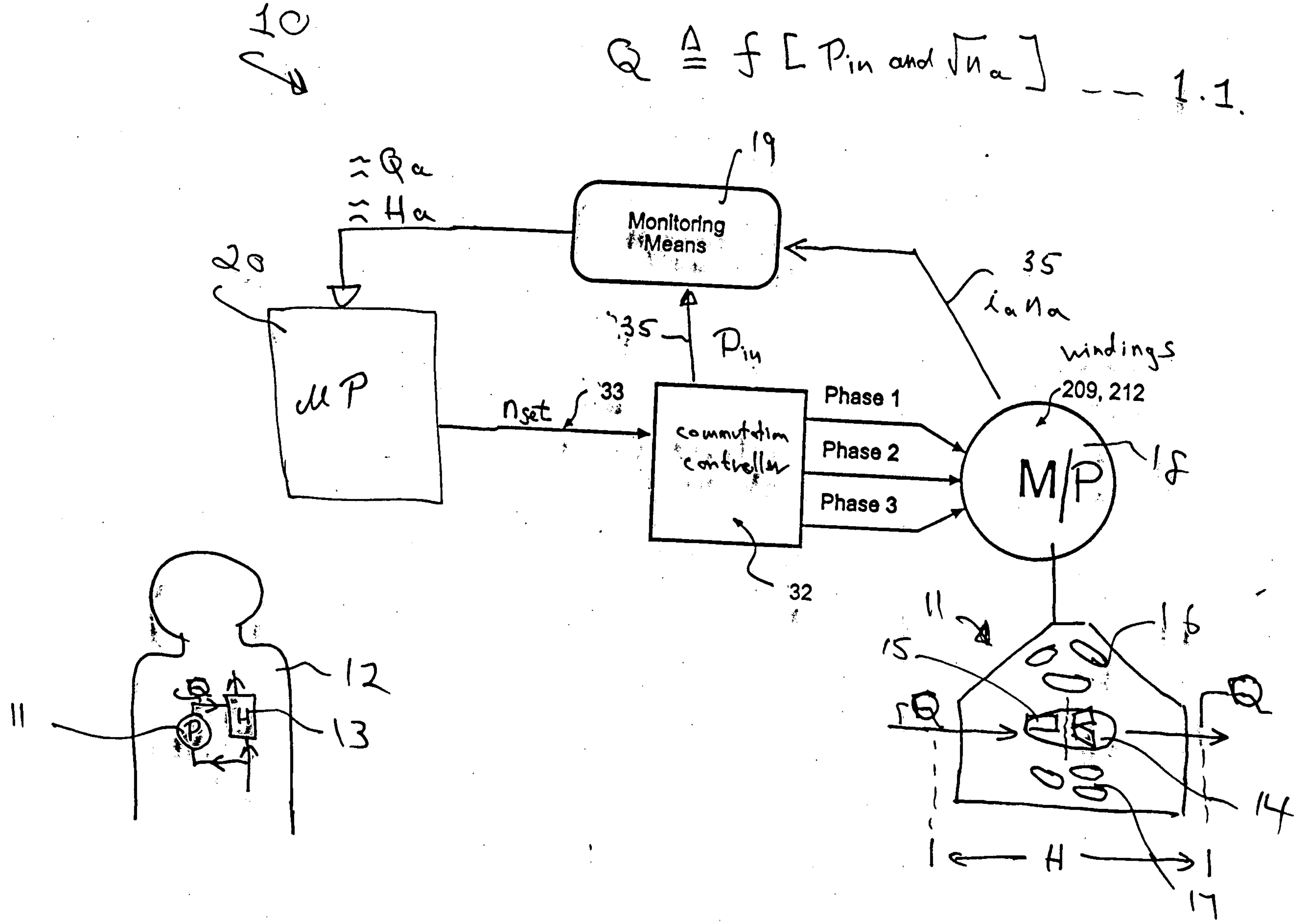



Fig 1

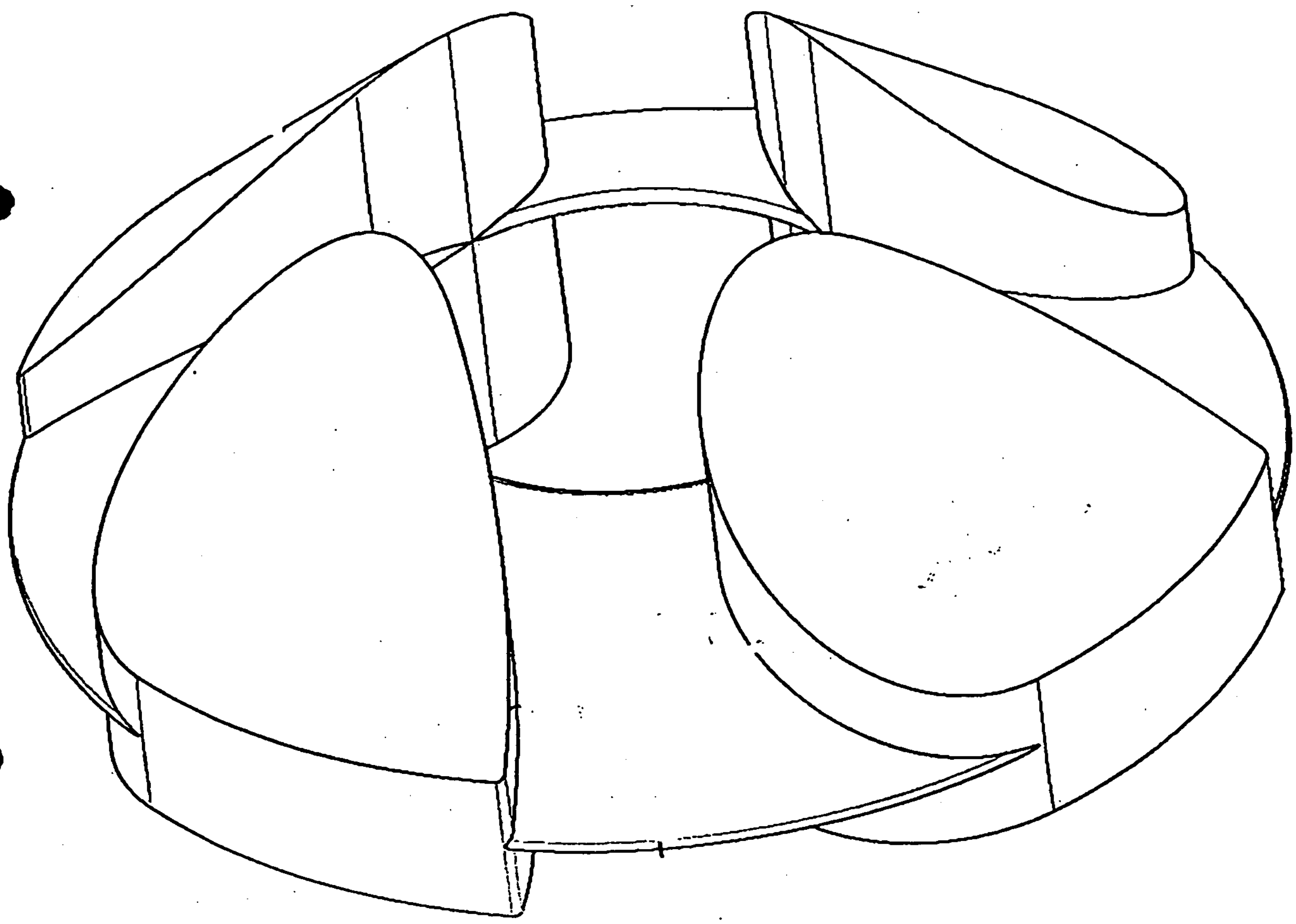


Fig 2

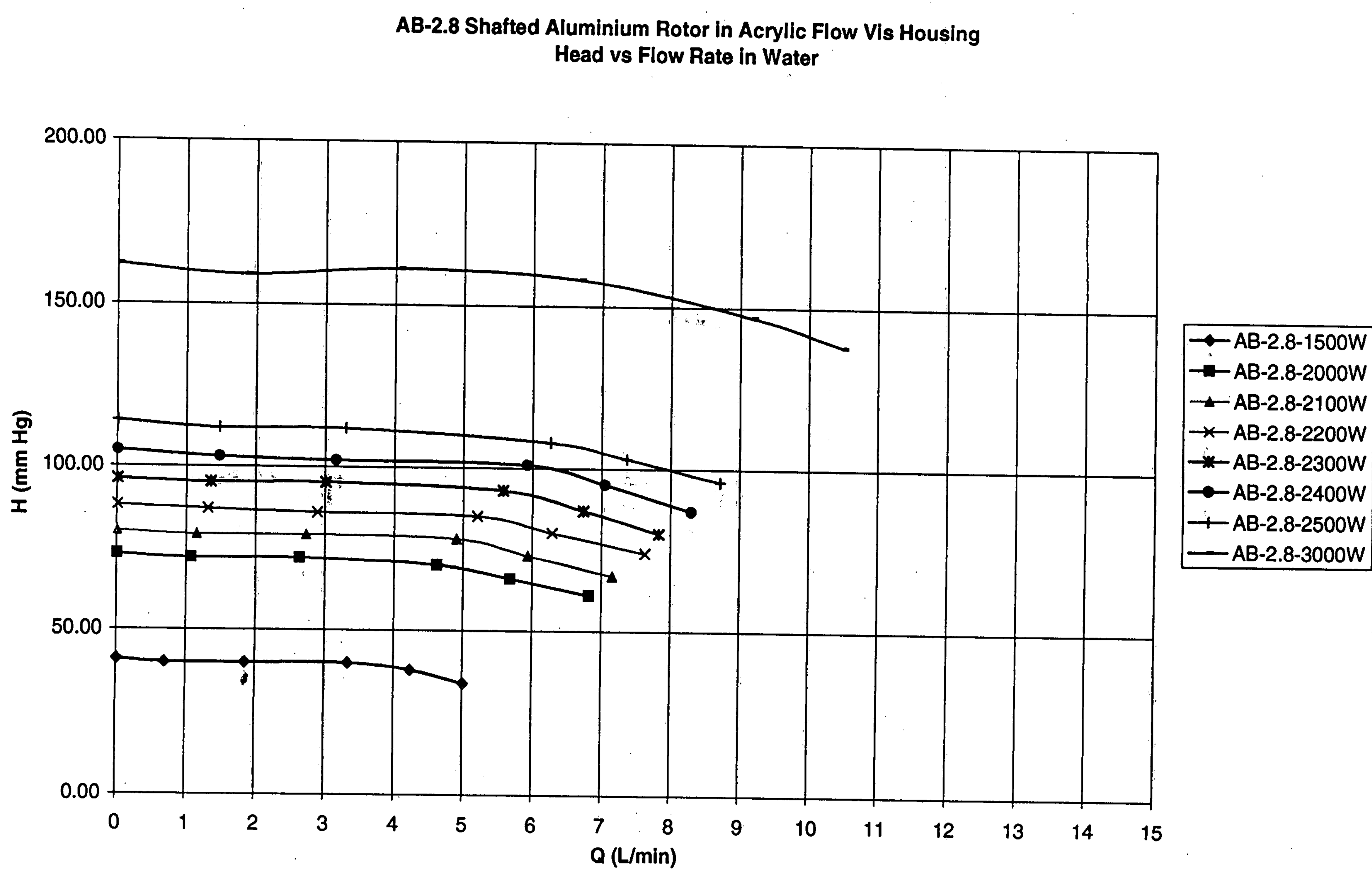

AB-2.8 Shafted Aluminium Rotor in Acrylic Flow Vis Housing
Head vs Flow Rate in Water
200.00
150.00
100.00
50.00
0.00
H (mm Hg)
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
Q (L/min)
AB-2.8-1500W
AB-2.8-2000W
AB-2.8-2100W
AB-2.8-2200W
AB-2.8-2300W
AB-2.8-2400W
AB-2.8-2500W
AB-2.8-3000W

Fig 3

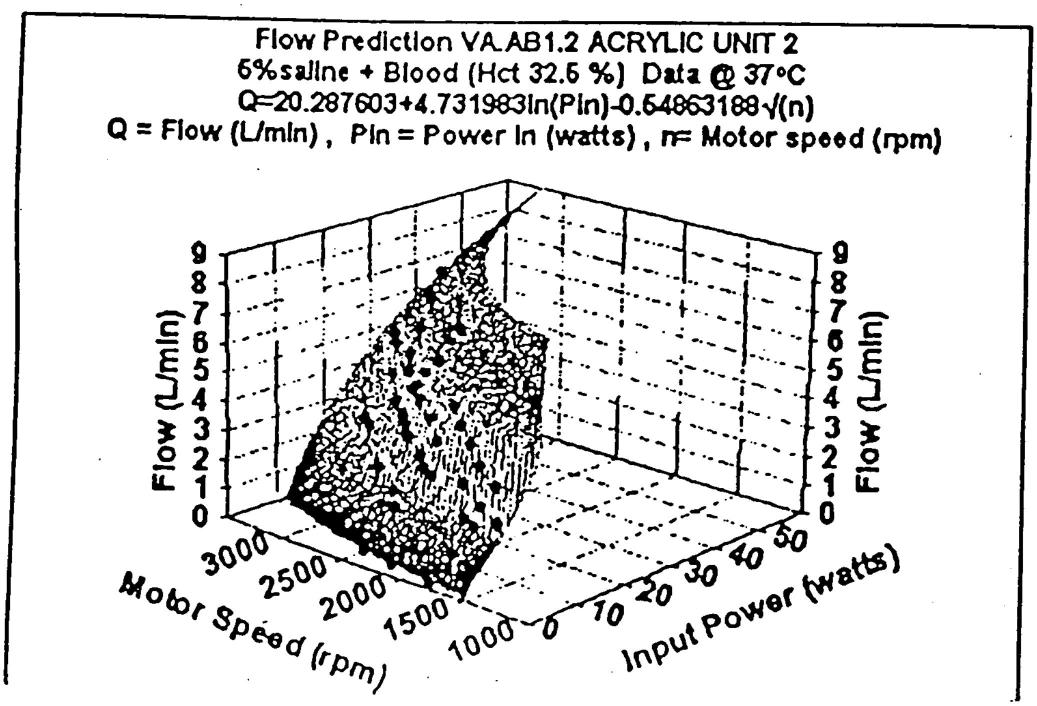

Flow Prediction VA.AB1.2 ACRYLIC UNIT 2
6%saline + Blood (Hct 32.5 %) Data @ 37°C
Q=20.287603+4.731983ln(Pin)-0.54863188√(n)
Q = Flow (L/min) , Pin = Power In (watts) , n= Motor speed (rpm)
Flow (L/min)
Motor Speed (rpm)
Input Power (watts)
3000
2500
2000
1500
1000
0
10
20
30
40
50
0
1
2
3
4
5
6
7
8
9

Fig 4

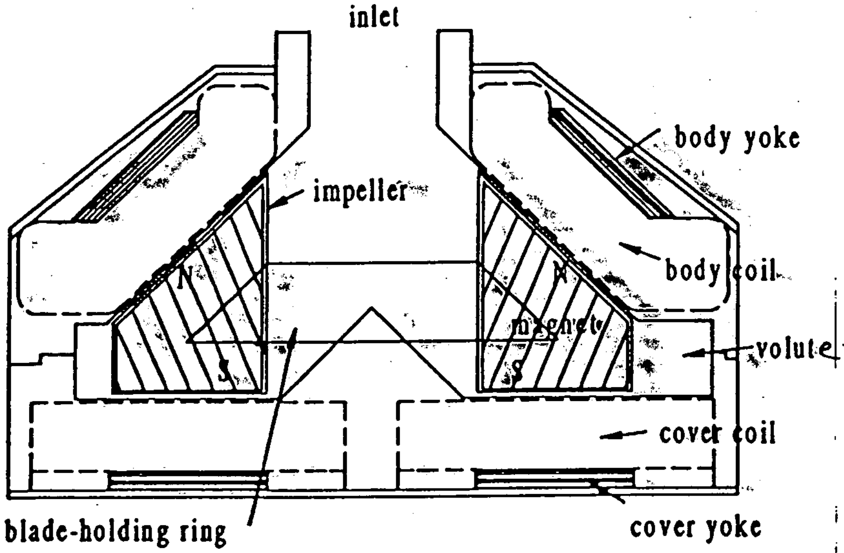
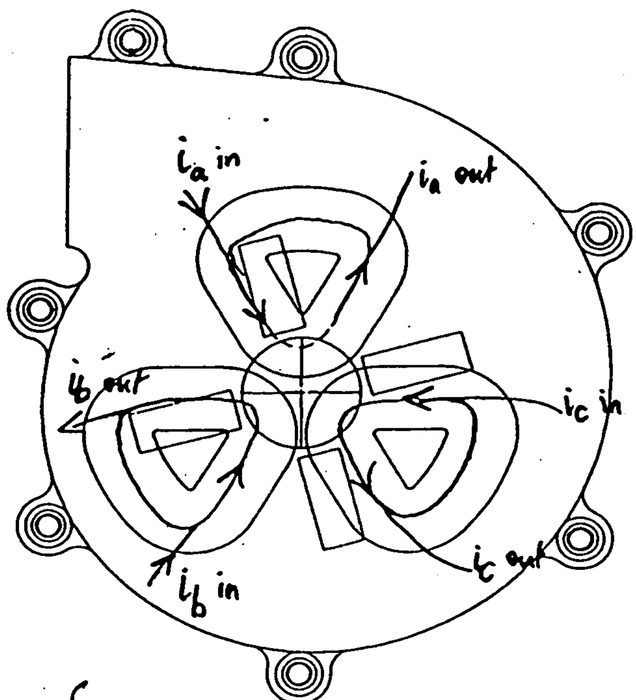



Fig. 5 Sketch of the pump cross-section.

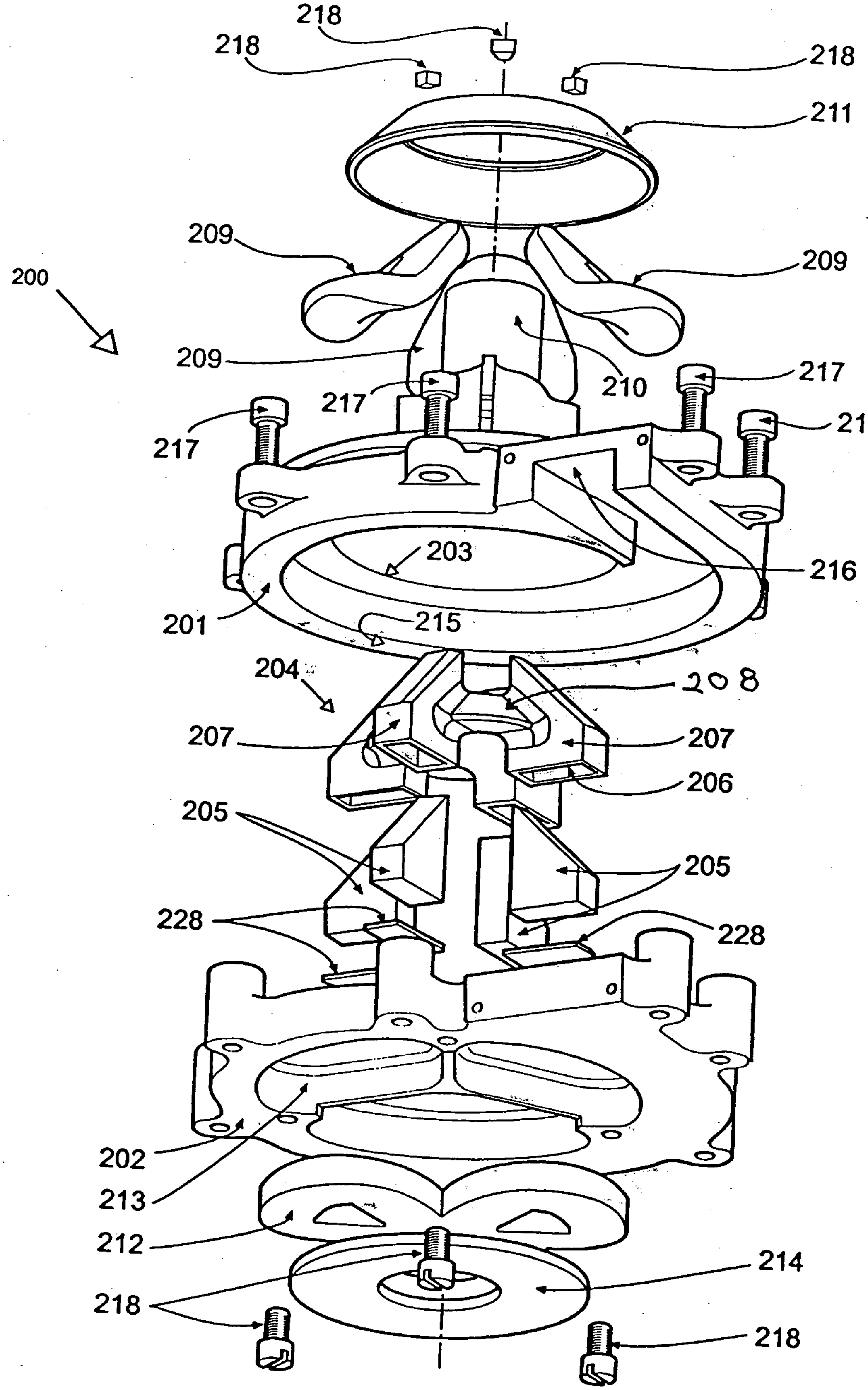
Fig 5




6

Fig. . Cover coil superimposed by rectangular magnets.

Fig 6



218
218
218
211
209
209
200
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205
228
228
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213
212
214
218
218


Fig. 7

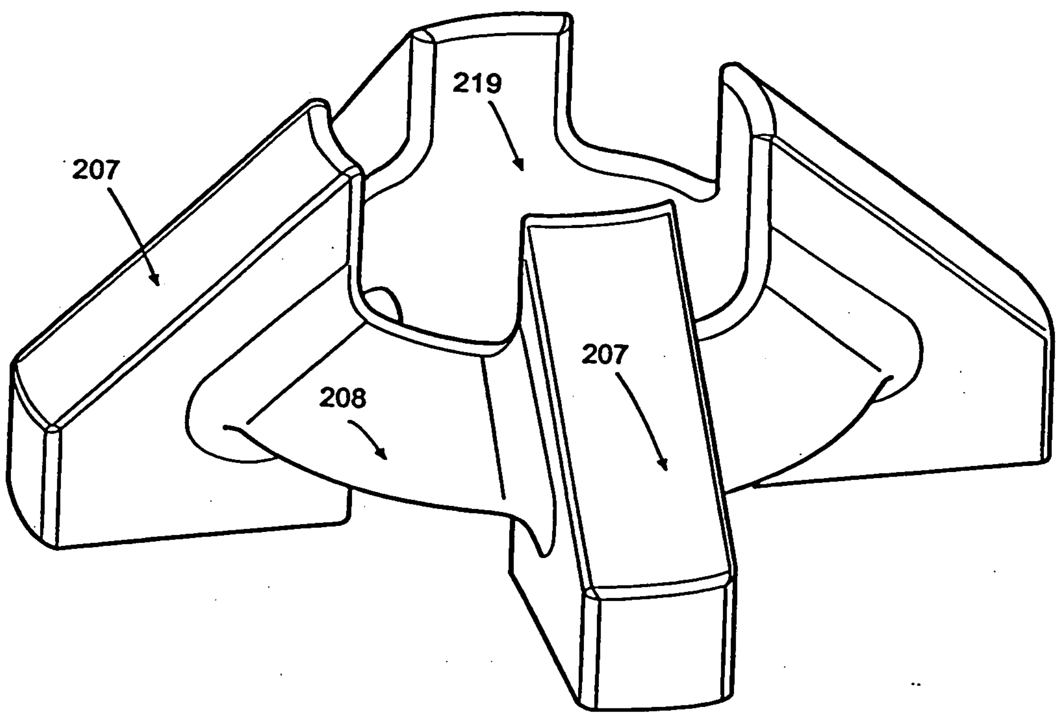



Fig. 8

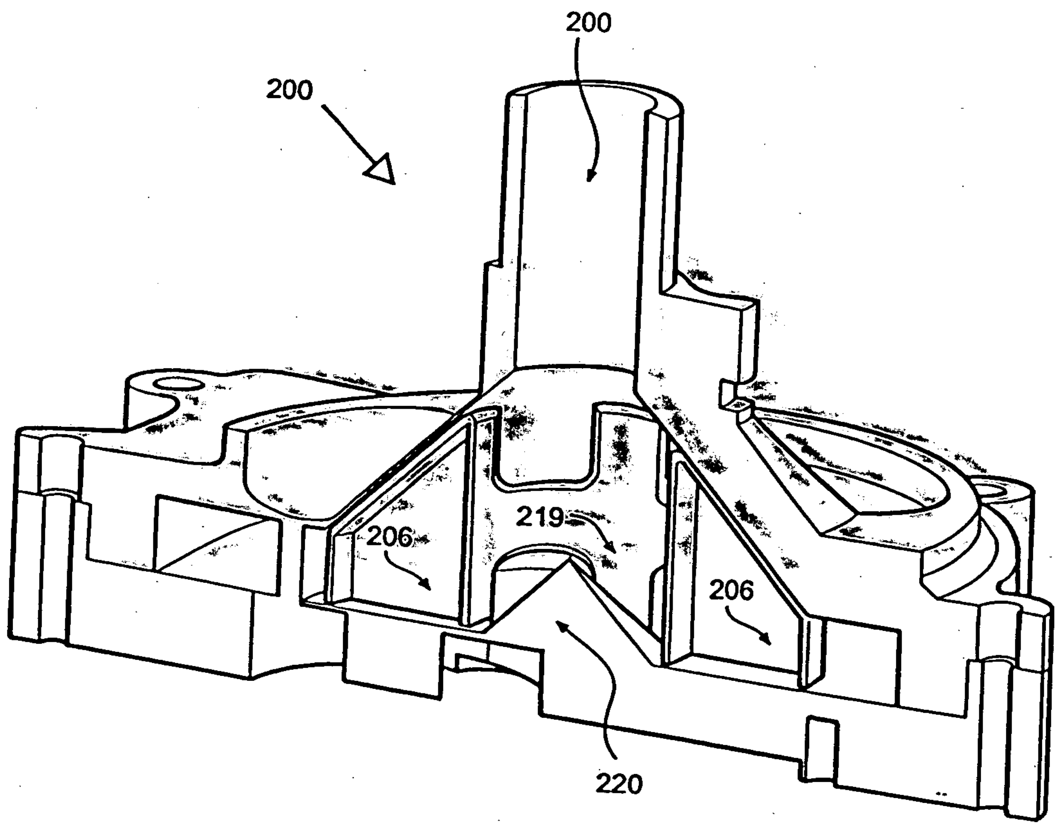



Fig. 9

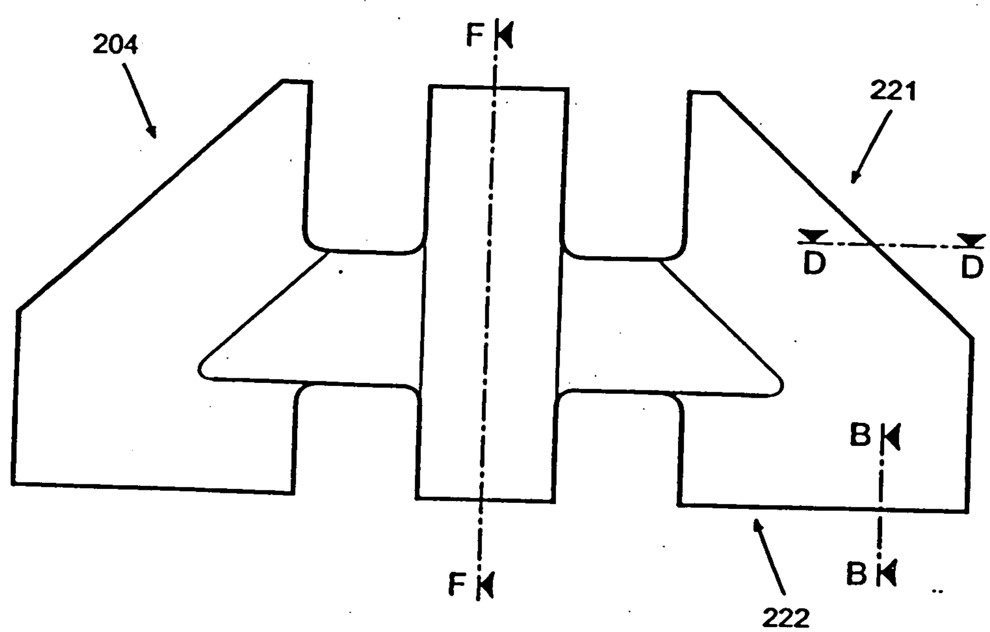



Fig. 10

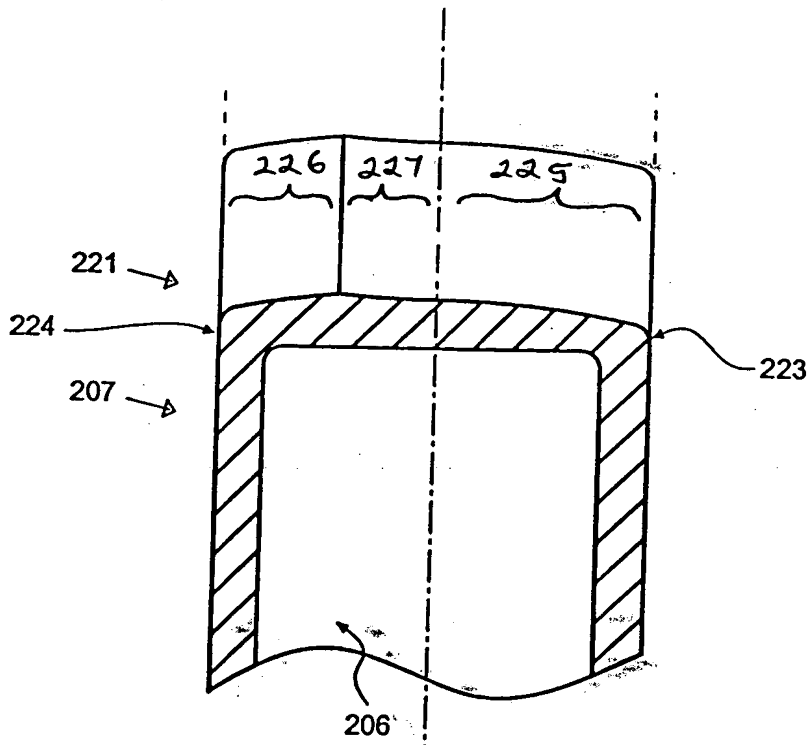



Fig. 11 A

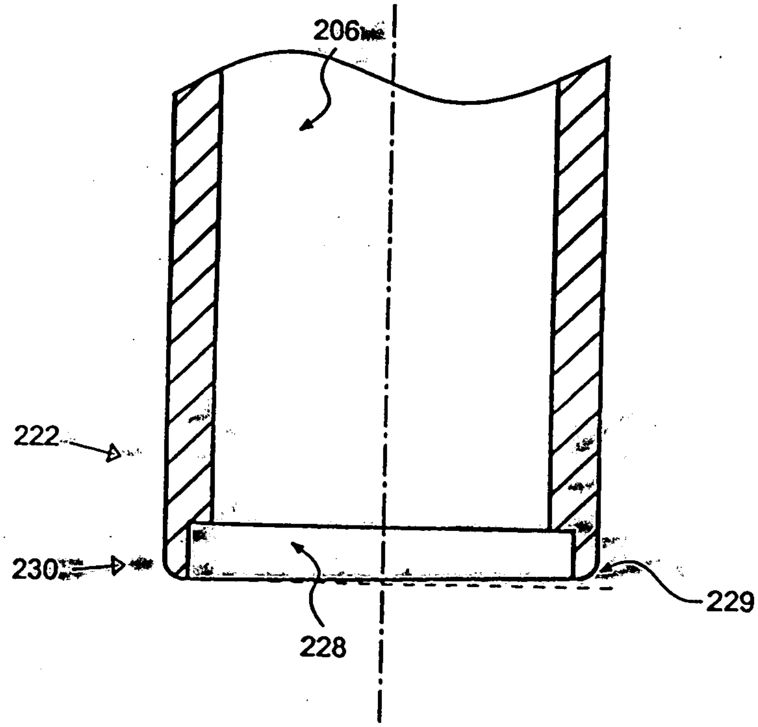



Fig. 11 B

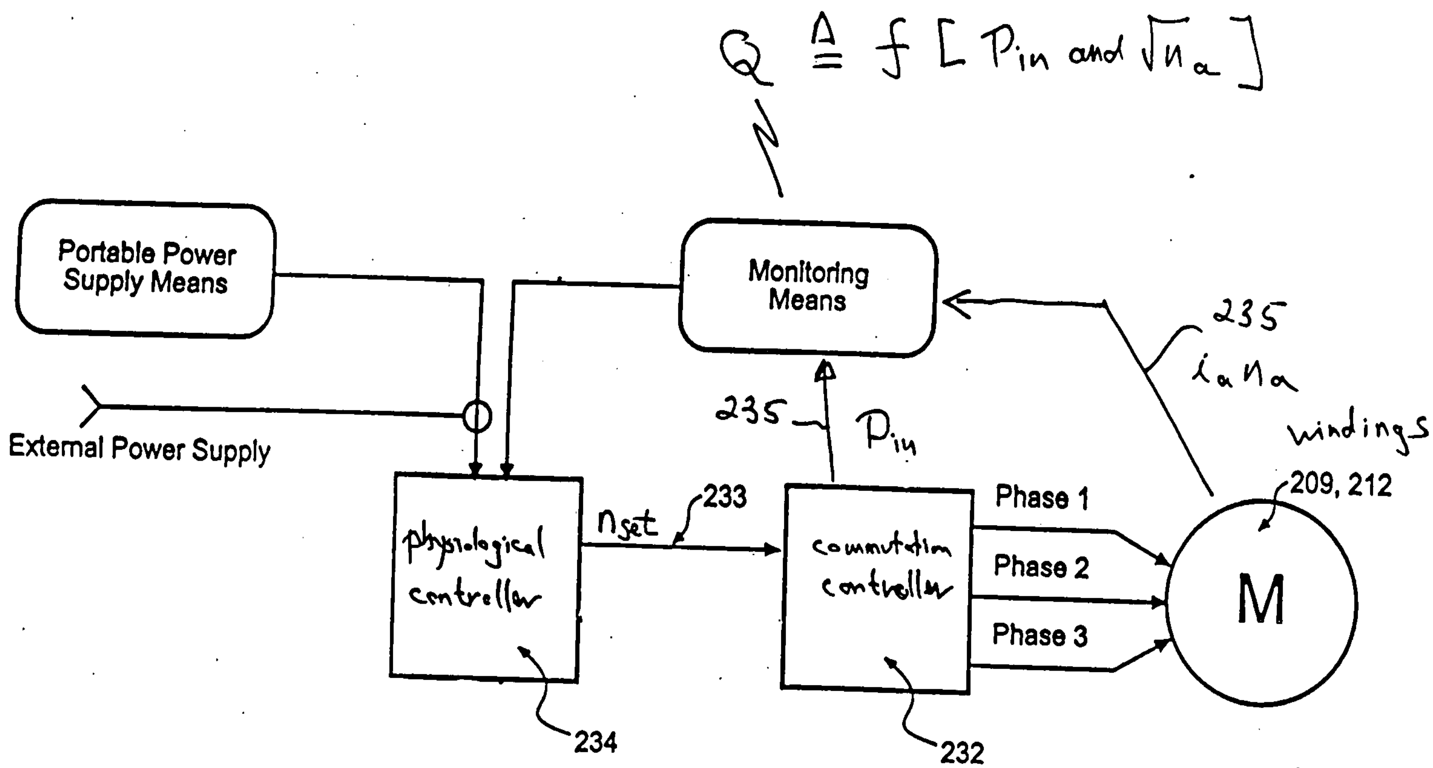

$Q \triangleq f [ P_{in} \text{ and } \sqrt{n_a} ]$
Portable Power Supply Means
External Power Supply
Monitoring Means
235
$i_a n_a$
windings
209, 212
M
235
$P_{in}$
physiological controller
$n_{set}$
233
commutation controller
Phase 1
Phase 2
Phase 3
234
232

Fig. 12

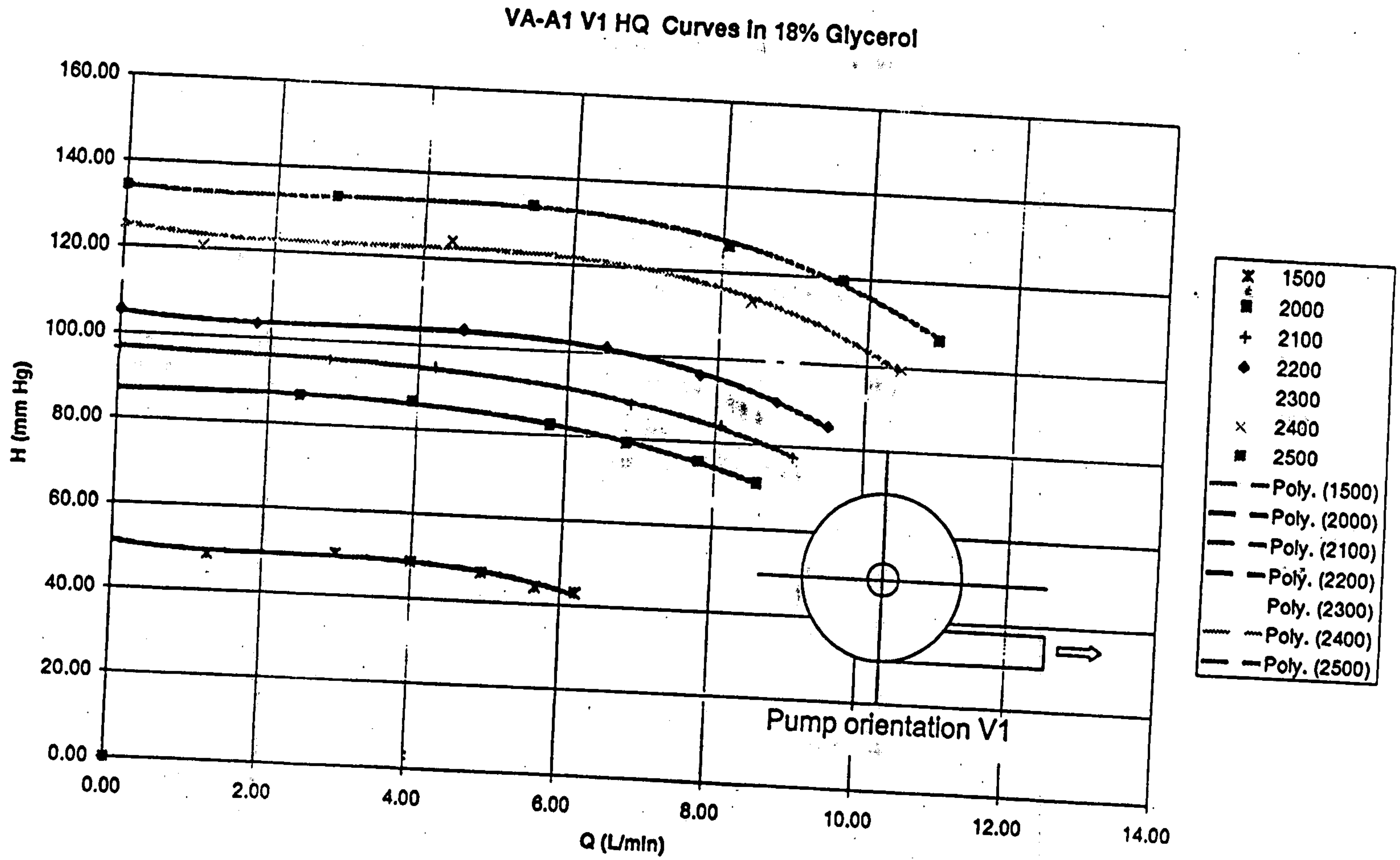

VA-A1 V1 HQ Curves In 18% Glycerol
160.00
140.00
120.00
100.00
80.00
60.00
40.00
20.00
0.00
H (mm Hg)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
Q (L/min)
Pump orientation V1
1500
2000
2100
2200
2300
2400
2500
Poly. (1500)
Poly. (2000)
Poly. (2100)
Poly. (2200)
Poly. (2300)
Poly. (2400)
Poly. (2500)

Fig. 13

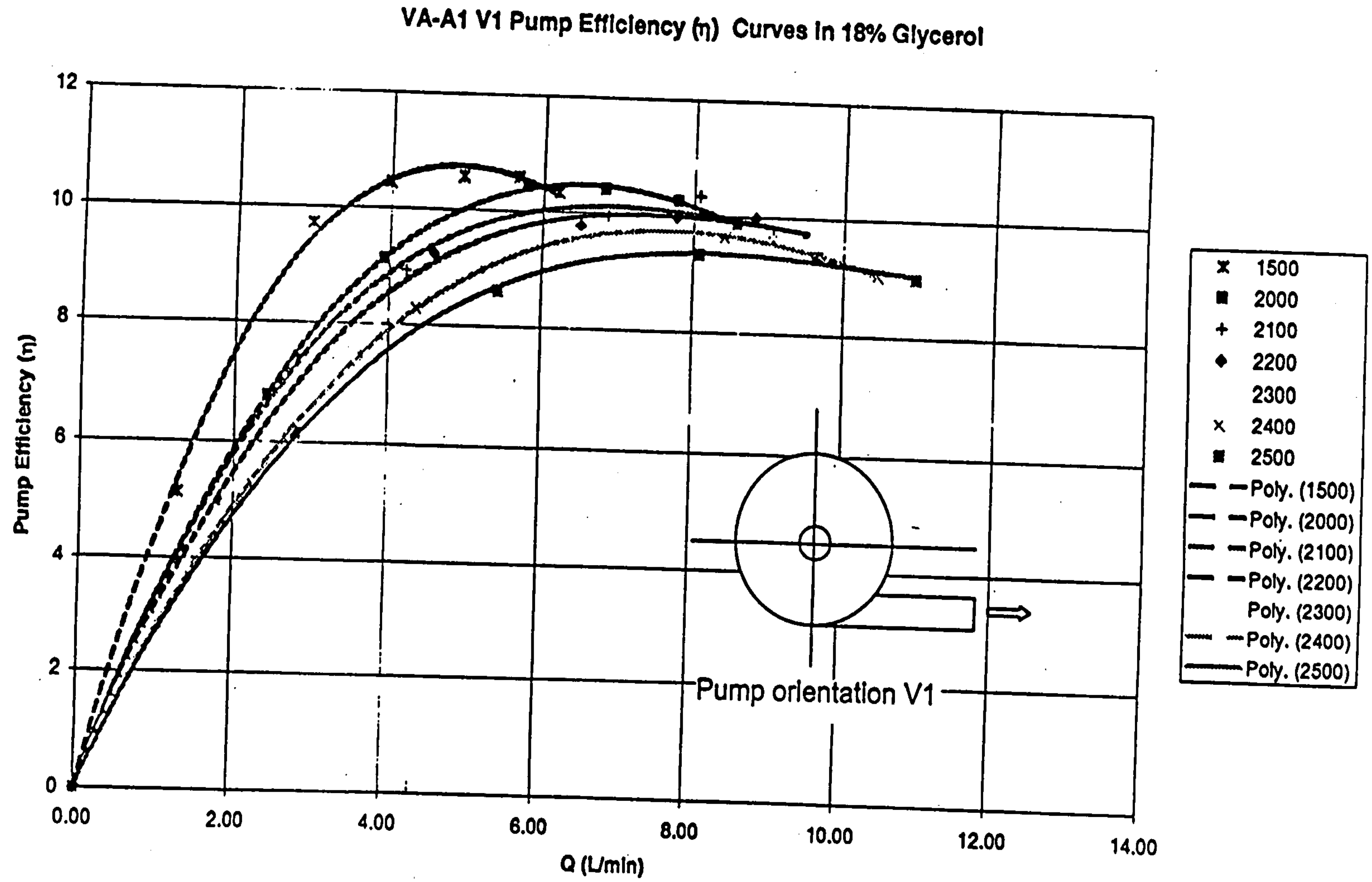

VA-A1 V1 Pump Efficiency (η) Curves in 18% Glycerol
Pump Efficiency (η)
Q (L/min)
0
2
4
6
8
10
12
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
1500
2000
2100
2200
2300
2400
2500
Poly. (1500)
Poly. (2000)
Poly. (2100)
Poly. (2200)
Poly. (2300)
Poly. (2400)
Poly. (2500)
Pump orientation V1

Fig. 14

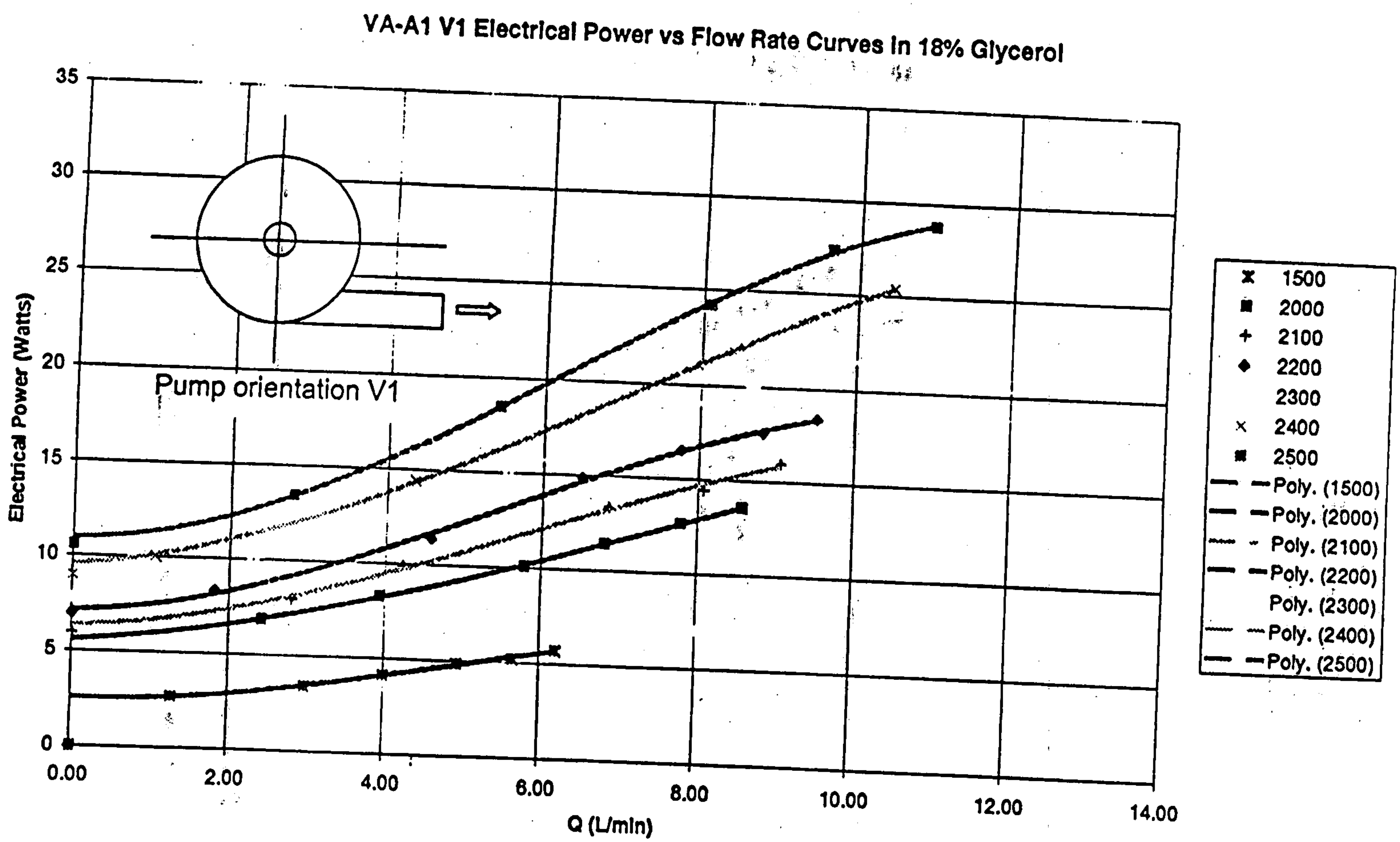

VA-A1 V1 Electrical Power vs Flow Rate Curves In 18% Glycerol
Electrical Power (Watts)
35
30
25
20
15
10
5
0
Q (L/min)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
Pump orientation V1
1500
2000
2100
2200
2300
2400
2500
Poly. (1500)
Poly. (2000)
Poly. (2100)
Poly. (2200)
Poly. (2300)
Poly. (2400)
Poly. (2500)

Fig. 15

16/19

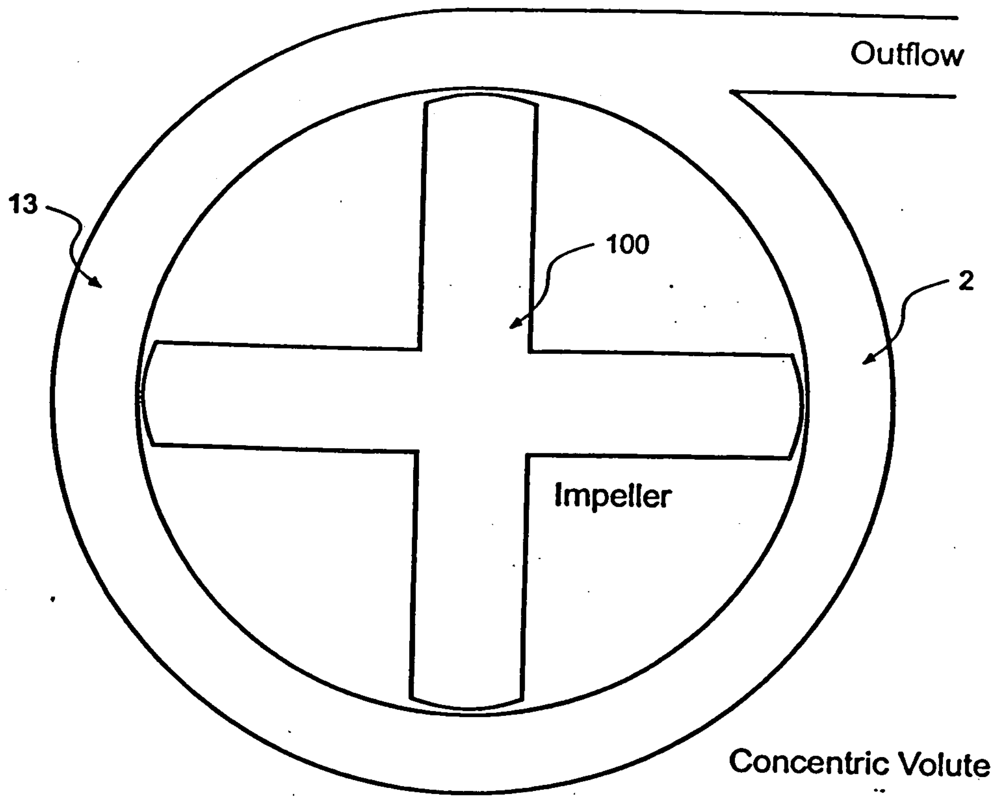



Fig. 16

17/19

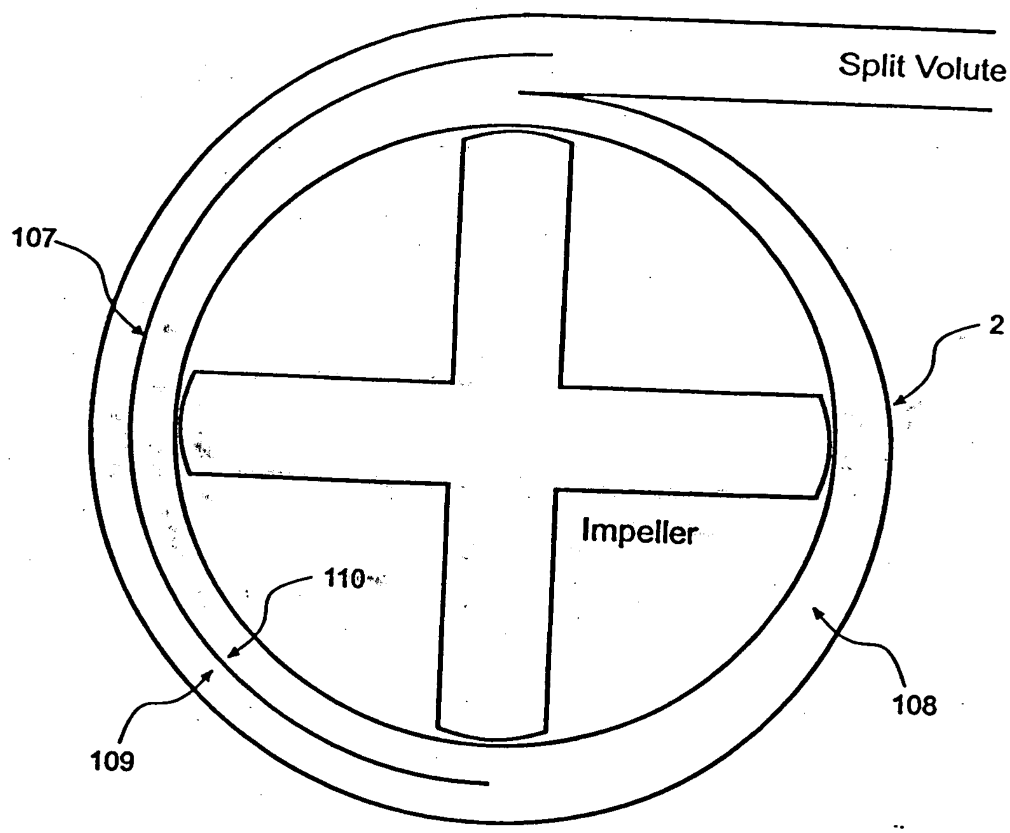



Fig. 17

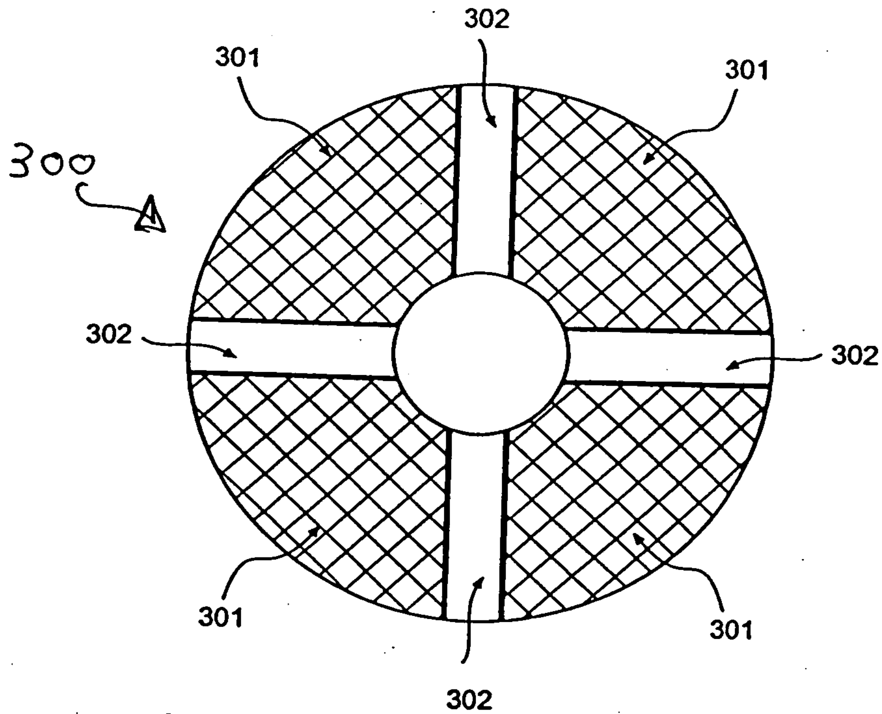

302
301
301
300
302
302
301
301
302

Fig. 18

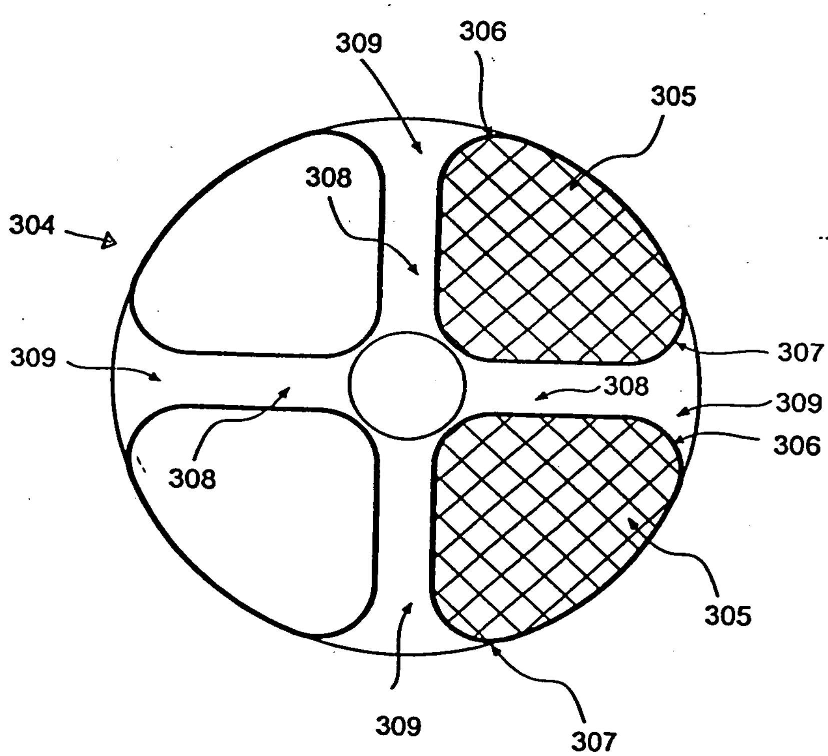
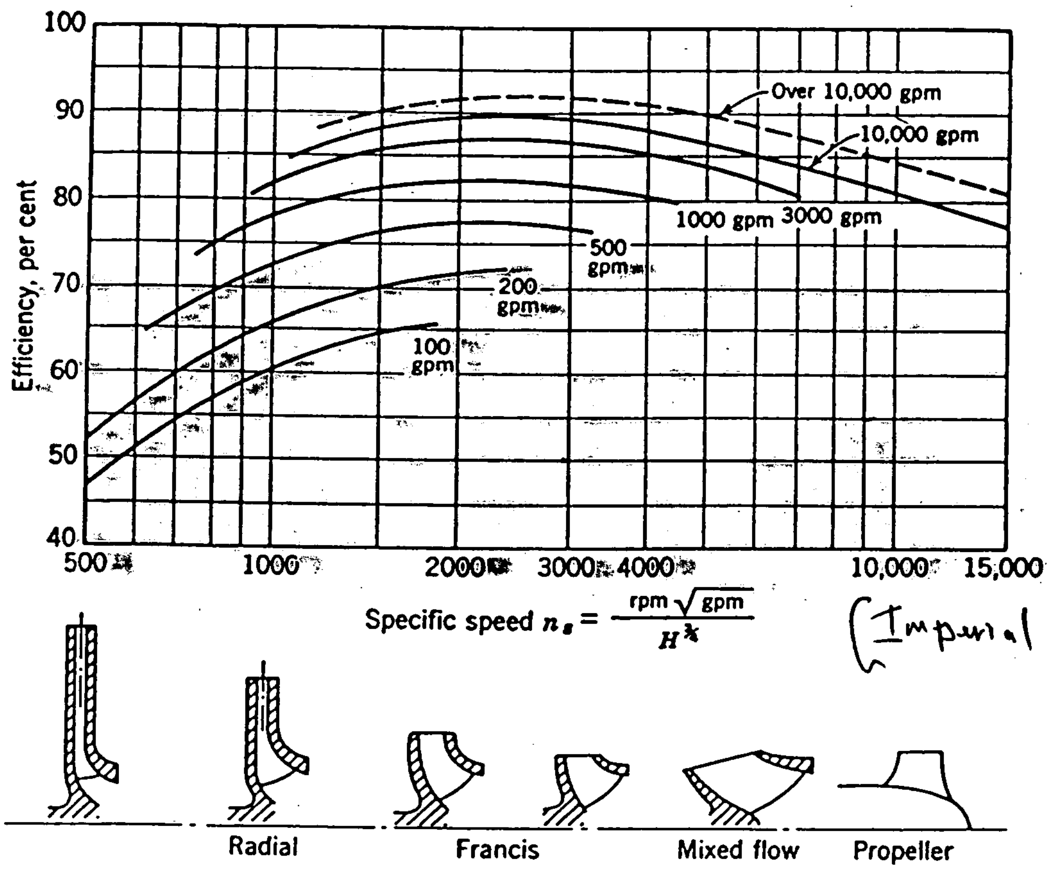

309
306
305
308
304
309
307
308
309
308
306
305
309
307

Fig. 19




[Imperial units]

Pump efficiency versus specific speed and pump size (Worthington).

From "Flow-Pumps - Design + Application"
Published by Krieger Publishing Co, Fl, USA. 1993.

Fig 20